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DESIGN SIMULATION TO PASS TESTS FOR ACCEPTANCE AS DEFINED IN prEN14363 OR UIC518

Щоб бути упевненим, що знову розроблений і створений рейковий екіпаж успішно пройде приймальні випробування, необхідно досліджувати його поводження за допомогою комп'ютерного моделювання. Оцінка впливу зміни різних параметрів дозволяє також оптимізувати розміри і параметри конструкції. У роботі детально описані необхідні для проведення оцінки динамічних якостей екіпажа варіанти моделювання, показано, які параметри визначають точність рішення поставленої задачі.

Чтобы быть уверенным, что вновь разработанный и созданный рельсовый экипаж успешно пройдет приемочные испытания, необходимо исследовать его поведение с помощью компьютерного моделирования. Оценка влияния изменения различных параметров позволяет также оптимизировать размеры и параметры конструкции. В работе детально описаны необходимые для проведения оценки динамических качеств экипажа варианты моделирования, показано, какие параметры определяют точность решения поставленной задачи.

Introduction

International rail traffic is possible due to international standards and acceptance procedures. To reduce the needed time and money to pass the tests design calculations are very helpful. Powerful rail vehicle calculation tools as ADAMS/Rail, MEDYNA, SIMPACK, VAMPIRE and others were developed and verified during the last years.

More important than MBS-algorithms is a realistic modelling of design elements as spring, damper, bump stop, friction element with their non-linear characteristics and last but not least modelling of wheel rail contact. An accurate result is more important than savings in calculation time.

The standards represent the experience of European railways in testing for acceptance of the running characteristics of vehicles. Simplified methods are also included to reduce expense of tests. This paper deals with the main aspects only of limits and procedures given in the standards.

Aim of Design Optimisations

The aim of a design is to build a vehicle with a good performance at a low price. Good running behaviour and low life cycle costs are important aspects for a decision about design parameter. Forces between wheel and rail, wear of profiles, comfort in the car body has to be optimised running on straight tracks with maximal velocity and through curves with high or low cant deficiencies.

Full tests, partial tests and simplified measurement method

The necessary amount of tests, methods and signals which have to be measured is depending on characteristics of the vehicle as axle load, velocity and knowledge of the design. Simulation allows investigating the dynamic behaviour and all values of the vehicle in detail. This opportunity should be used to find a good solution during the design process. The possibility of simplified measurement methods are not dealt with further as design calculations should not be restricted to these values. Information and knowledge about the vehicle performance are as well less with fewer values evaluated.

Assessment values

Only the assessment values of the normal measurement procedure are mentioned in the following. The values looked for are sorted according the reason as:
– running safety;
– track load;
– ride characteristics.

Limit values of running safety

Sum of guiding forces $\sum Y_{max}$ is the safety-critical limit for track shifting. The given limits are for a standard ballasted track with timber sleepers.

$$\Sigma Y_{max,lim} = k_1 * (10 + 2Q_o / 3) \text{ in kN, } (1)$$

$2Q_o$ is the static wheelset load in kN. The factor k_1 is depending on the type of vehicle:

$k_1 = 1.00$ for locomotive, multiple units and, passenger coaches,

$k_1 = 0.85$ for freight wagons.

The value of axle box force H is equivalent to $\sum Y$ but low pass filtered due to unsprung mass and primary suspension.

Quotient of guiding force and wheel force $(Y/Q)_{max}$ is the safety-critical limit of derailment for the quotient of a leading wheel.

$$(Y / Q)_{max,lim} < 0.8 \qquad (2)$$

on track.

There is no way to calculate Y from the H force measurement as there are wheelset internal forces as result of lateral friction.

Instability criterion is described with

$$\Sigma Y_{rms,lim} = \Sigma Y_{max,lim} / 2 . \qquad (3)$$

The random mean square value of a section has to be less than half of the track shift limit.

Limit values of track loading

The quasi-static guiding force is limited to

$$Y_{qst,lim} < 60 \text{ kN} \qquad (4)$$

and the quasistatic wheel force is limited to

$$Q_{qst,lim} < 145 \text{ kN} \qquad (5)$$

both excluding transition sections.

$$Q_{max,lim} = 90 + Q_o \text{ in kN} \qquad (6)$$

depending on maximum of permissible speed of the vehicle the force is limited to

$$Q_{max,lim} < 160 \text{ kN} \dots 200 \text{ kN} , \qquad (7)$$
$$v \quad < 160 \text{ km/h} \dots 300 \text{ km/h}. \qquad (8)$$

These values take account of rails with a weight per meter > 46 kg and the minimum value of rail strength of 700 N/mm².

Limiting values of ride characteristics

The limiting values of ride characteristics are accelerations in the vehicle body. They are measured in lateral and vertical directions in the centre and over the running gears. Maximal values and mean square values are limited depending on the type of vehicle.

Simulation of Stationary and 'On Track' Tests

Simulation makes it possible to study dynamic behaviour of the vehicle in detail. In simulation it is easy to define any kind of track and to analyse the vehicle for all track cases which have to be passed in future time.

Stationary Tests

The vehicle has to show in stationary tests that there is no risk of derailment, to exceed the kinematic envelope and loss of contact of the pantograph to the overhead wire.

Safety against derailment

The test conditions have been developed by ERRI and documented in several reports [3]. They are carried out on twisted test track with a radius of R = 150 m and changing cant from 45 mm to - 45 mm within 30 m to get a twist of +3 ‰.

The wheel with the lowest load has to run in leading position on the outer rail. The test twist is between 3 ‰ and 7 ‰ depending on the bogie wheel base and the distance $2a$ between the bogies.

$$g_{lim} = \min(7.0; 20/2a + 2.0) \qquad (9)$$

If test track twist is greater than 3 ‰ this have to be done using shims in the suspensions.

This test has to be calculated using time integration methods. The vehicle has to pass the test track with a speed lower than 10 km/h. Possible hysteresis within suspensions has to be modelled carefully. Rails must be in dry condition. Figure 1 shows the minimum of desired values of coefficient of friction $\mu = \tau$. Calculations should be done with higher coefficient of friction to simulate the worst case $\mu \sim 0.4\dots0.5$.

Safety against derailment as the worst case simulation

The tests conditions have been developed by British Rail and were partly used in the so-called Manchester benchmark track case 1 [6]. A linear dip of 20 mm over 6 m has to be put at the high rail in the run-off transition and a body-bogie yaw torque should act additionally.
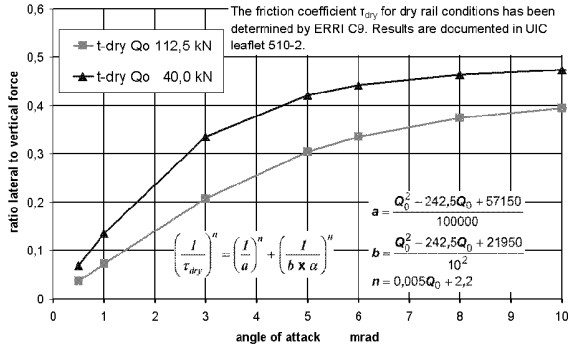

Fig. 1. Coefficient of friction measured by UIC [3]

Sway characteristic

The vehicles have to run inside a kinematic envelope and so they are not allowed to move freely. The test is done on a canted track to simulate curving, side wind etc. The calculation determines quasistationary equilibrium.

'On Track' Tests

On track tests investigate characteristics of a vehicle with regard to the interaction to the track layout and rail deviations. Test zones are track sections with the characteristic:

- straight;
- large curves R > 600 m;
- small curves 600 m > R > 450 m or
- very small curves 450 m > R > 250 m.

The length of sections is 70 m to 500 m according to track layout and speed.

Track geometry quality

The track quality is based on track maintenance criteria and a definition is given in the standard prEN14363:

a) quality level QN 1:
necessitates observing a track section or taking maintenance measures within the frame of normal operations scheduling,

b) quality level QN 2:
necessitates taking short-term maintenance measures,

c) quality level QN 3:

characterizes track sections which do not exhibit the usual track geometry quality. Quality level QN 3, however, does not represent the most adverse but still tolerable maintenance status.

The standards forced to do tests on tracks with quality QN 2 with 10 % of the total length of test sections. So this quality has mainly to be used for simulation as that is normally the most critical condition.

The quantities of track geometry deviations depend on line speed. The characteristic values are evaluated in a bandwidth of 3 m to 25 m wavelength. This is due to the standard performance of measurement devices of track geometry. But to get realistic simulation results the geometry has of cause to be used with a wider range of wavelength. The interesting excitation frequency normally is between 0.3 to 30 Hz. The conversion into wavelength L is done with the velocity.

$$L = f * v \qquad (10)$$

with f – frequency in Hz and v – velocity in m/s.

Definition of deviation with PSD

The dependency of amplitude from frequency or wavelengths Ω in rad/m is defined by a power spectral density PSD-function [5]:

$$S = \frac{A * \Omega_c^2}{(\Omega^2 + \Omega_c^2) * (\Omega^2 + \Omega_r^2)} \quad \text{in} \quad \frac{m^2}{rad/m} \quad (11)$$

with lateral A = 0,6125 and vertical A = 1,08 in $rad * m$ for high level and

Ω_c = 0,8426 and Ω_r = 0,0206 in $rad * m$.

Ω in $rad * m$

Example 1. Realisation of rail deviations, line speed 80 km/h.

The lateral displacements of right and left rails are shown along the track in figure 2. The values of the light grey curve are filtered in the range between 3 m to 25 m to get the quality level. The dark curve represents the values in a wavelength between 1 m and 70 m.

The spectrum shows that the range of measured bandwidth is 70 m to 3 m. The dark curve shows the measured values and the light grey curve are band

pass filtered values in a range of 25 m to 3 m wavelength in the plot over the distance in Figure 3 [4].
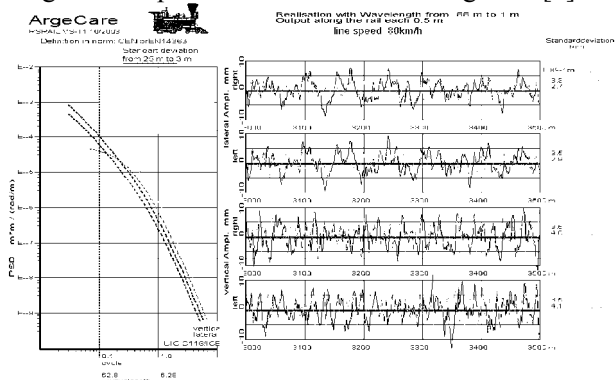

Fig. 2: Realisation of rail deviations [4]

Example 2. Evaluation of measured track with line speed 250 km/h.
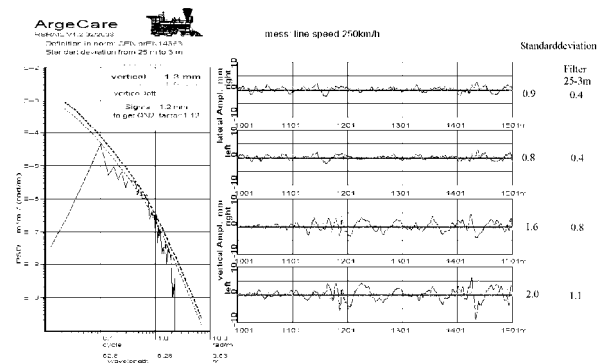

Fig. 3: Evaluation of measured rail deviations [4]

Conicity, wheel rail profiles

The standard gives only some hints about profiles of wheel and rail. The running characteristics of a vehicle are influenced by the combination of wheel and rail profiles. One describing function is the so-called conicity. This is a function of the difference of left and right wheel roll radii at lateral amplitudes. The prEN 14363 defines the so called equivalent conicity $\tan(\gamma_e)$.

For a given wheelset running on given track it equals the tangent of the taper angle of a tapered profile wheelset whose transverse movement has the same wavelength of kinematic yaw as the wheelset under consideration.

The wheel profiles of a wheelset are combined with rail profiles (including the inclination) and the track gauge.

Conicity values are found on freight lines up to $\tan(\gamma_e)$ =0.8 due to pure maintenance or narrow gauge. And on high speed line there is a conicity up to $\tan(\gamma_e)$ =0.3 or even 0.4.

The standard defines to measure wheel profile in service to analyse if conicity has changed. Calculations are necessary to derive if profile wear changes conicity during service as shown in Figure 4.

Design simulations

To design a vehicle with good performance and to know test results in advance a lot of computer simulations have to be done.
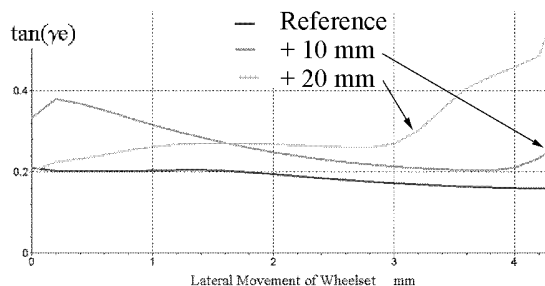

Fig. 4: Change of conicity due to profile wear (RSGEO)

Natural behaviour

The natural behaviour of a rail vehicle depends on the speed it is running. Hunting mode and frequency depends on the velocity of the vehicle. Some of the eigen-modes change with speed quite a lot, other only slightly. The ratio of damping and frequency of some modes are shown in Figure 5. If damping falls below a certain value (in theory zero) the so-called critical speed is reached.
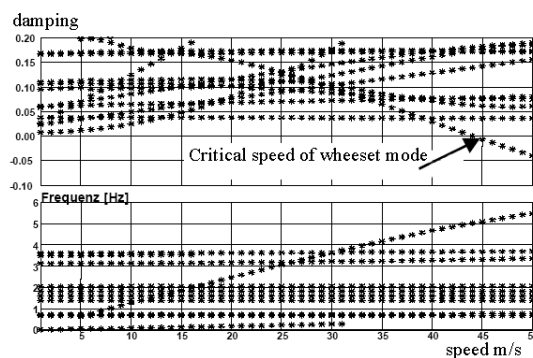

Fig. 5: Damping and frequency depending on speed

Figure 6 shows as function of speed the frequencies of some body modes and the speed depending hunting mode.

Evaluation of Stability

Hunting phenomena occurs due to selfexciting effects in wheel rail contact. Below the so-called critical speed movements of wheelsets are damped; above they are increasing with the behaviour of non-linear systems to approach a limit cycle.
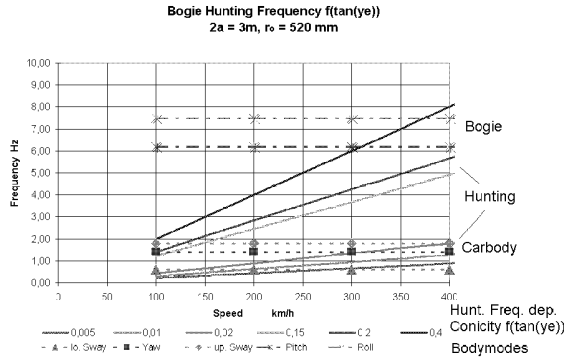

Fig. 6: Frequencies as a function of speed

Rail irregularities will normally stabilize the system. Which means calculated critical speed on smooth track is on save side for the system.

This limit depends on:
– coefficient of friction between wheel and rail;
– conicity;
– suspension parameter of the vehicle;
– masses, etc.

Therefore an evaluation is done to find the speed depending eigen-mode which has the lowest damping ratio. If an additional parameter is changed lowest ratios damping ratio can be shown in a 3D-plot or contour plot – stability chart. Lines of same damping are plotted against conicity $\tan(\gamma_e)$ and speed in Figure 7.

Beside conicity other parameters of the vehicle can be changed as e.g. lateral primary stiffness of wheelsets.

Another method to determine critical speed is to investigate self-excited movement of the vehicle. A non-linear integration is done with changing velocity. At high speed hunting mode is initiated either by self or forced excitation. The hunting movement which can show different shapes as velocity is decreased must die out. When altitudes on undisturbed track become zero, the speed is called critical speed. The critical speed of two design variants is shown in Figure 8.
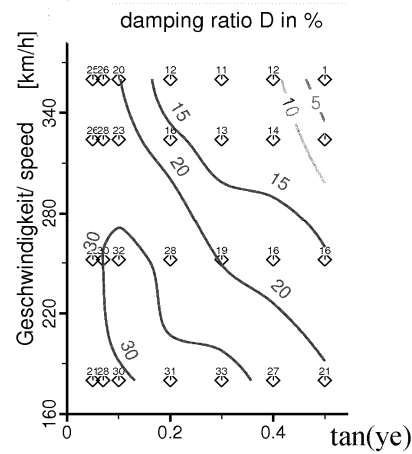

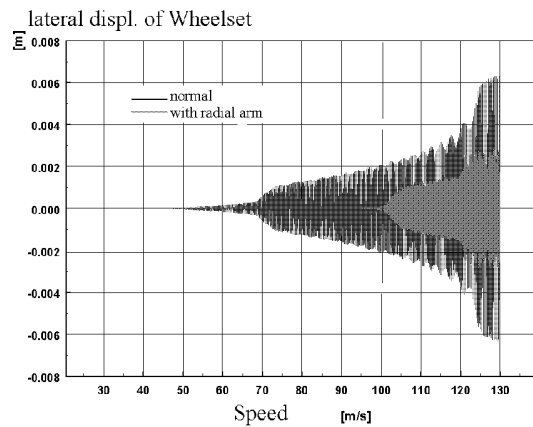Fig. 7: Results of linear stability simulation

Fig. 8: Results of non-linear stability simulation (MEDYNA)

The critical speed rises from 50 to 95 m/s using an improved bogie design fitted with a radial arm.

Vehicle interaction with track

The running behaviour describes the characteristics of a vehicle with regard to the interaction between vehicle and track. Track layout is defined as:
– straight track with variation of speed;
– curved track;
– large curves with speed up to maximal velocity;
– small and very small curves with unbalanced lateral accelerations up to maximal cant deficiency.

Rail deviations have to be selected according the speed.

All these simulations should be done with variations of:
– coefficients of friction;

– wheel and rail profiles;
– worn profiles / change in conicity.

Behaviour of vehicle with very strong side wind

High-speed vehicle get in trouble running through very high side dusts. This occurs running on embankments or over bridges. Such situation has to be investigated calculating such situations with lateral forces acting on the car body.

Optimisation of vehicle performance

The performance of rail vehicle mainly depends on:
– stiffness of primary and secondary suspensions,
– damping parameter,
– parameter of additional devices to get better stability or curving e.g. using the radial arm design of Dr. Scheffel.

A design optimisation is shown using example of a bogie with radial arm. Influence of shear stiffness on damping is shown in Figure 9.
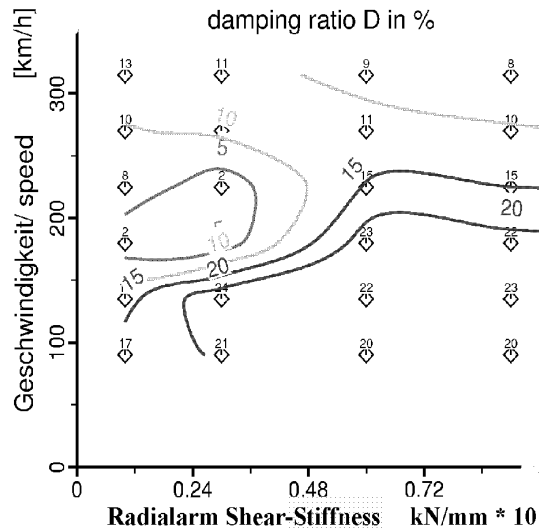
Fig. 9: Parameter variation

Calculation of wear

Results of a wear calculation are shown in Figure 10. A light rail vehicle runs through a very narrow curve and wear of the rails is calculated.

Use of simulation results during on track test

Another possibility is to use results of simulations to verify the measurement equipment starting

up evaluation tests.
– correct sign of signals;
– amplifier settings and
– …

Running through a switch gives a good overview about the measured signals as movements and forces are large and well known.
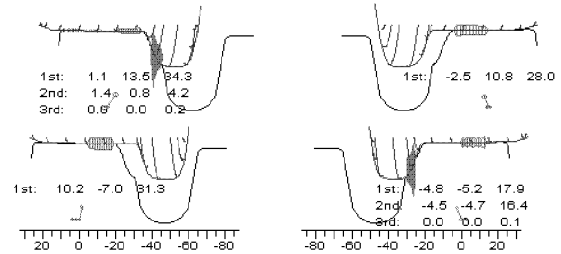
Fig. 10: Wear of rails caused by a light rail vehicle

Integration methods and data sampling

There is a conflict between needed computer time and storage and the accuracy of results. To solve very stiff algebraic equation leads to some numerical problems. During quite short time steps the integration routines calculate very high values so called spikes. They depend on chosen constant time step or accuracy defined in variable time step methods. Sampling method used in simulation should be similar to the method used in test runs. The data sample is stored at discrete time steps. Integration is done with much smaller steps. Therefore the series of calculated values of each integration step have either to be low pass filtered or a mean value has to be calculated over the period of output time steps.

Processing of calculated values

The calculated results must use the same signal processing procedure as doing measurements:
– sampling rate;
– filtering;
– method of classification;
– characteristic values: frequency-, rms- , mean- or max-values.

The statistical methods used in measurements to estimate maximum value of the samples of all measured section can not be used in calculations as there are too few results. So the maximum of a assessment value out of all calculations are used as estimated value and compared to the limit value.

$$\lambda = V_{\max} / V_{\lim} * 100 \quad \text{in} \% \qquad (12)$$

The results of each wheel and of each evaluation section are shown.

Conclusion

Computer simulations promote designing a vehicle with good running performance and to prevent to fail in an assessment test. Simulation makes it possible to study dynamic behaviour of the vehicle in detail. This opportunity should be used to find a good solution during the design process. In simulation it is easy to define any kind of track and to analyse the vehicle for all operation conditions.

A condensed form of assessment values is shown in Figure 11.

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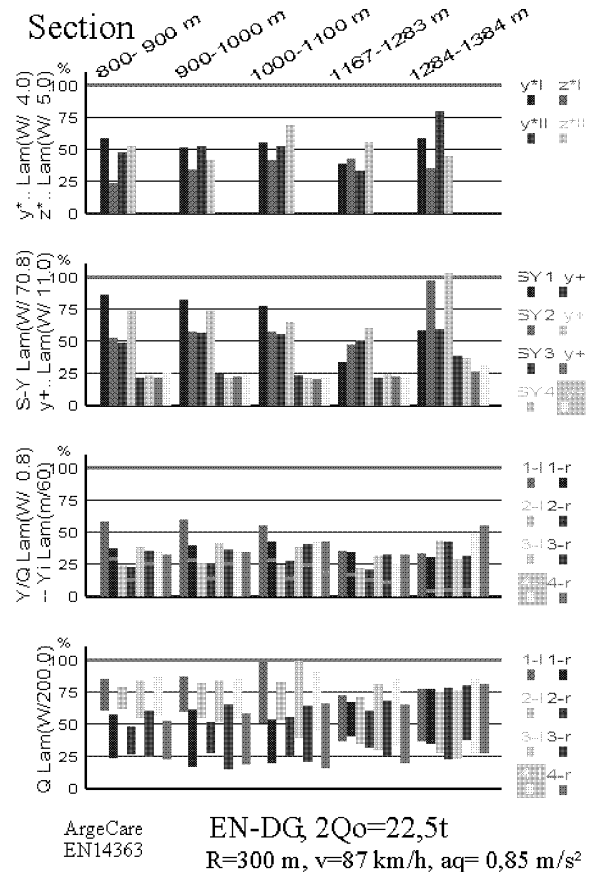6. COMPUTER SIMULATION OF RAIL VEHICLE DYNAMICS' Vehicle System Dynamics Vol. 30, Numbers 3-4, September 1998.

Fig. 11: Results in compact presentation